EXPANSION VALVE

Cross-References to Related Applications, If Any:

This application claims priority of Japanese Applications No.2002-202013 filed on July 11, 2002, entitled "Expansion Valve" and No.2003-133266 filed on May 12, 2003, entitled "Expansion Valve".

BACKGROUND OF THE INVENTION

(1) Field of the Invention

This invention relates to an expansion valve, and more particularly to a thermostatic expansion valve in a refrigeration cycle of an automotive air-conditioning system, for allowing a high-temperature and high-pressure liquid refrigerant to expand into a low-temperature and low-pressure refrigerant to supply the refrigerant to an evaporator, and at the same time controlling a flow rate of the refrigerant such that the refrigerant at an outlet of the evaporator is placed in a predetermined degree of superheat.

(2) Description of the Related Art

In the automotive air-conditioning system, a refrigeration cycle is formed in which a high-temperature and high-pressure gaseous refrigerant compressed by a compressor is condensed in a condenser, and an expansion valve allows the condensed liquid refrigerant to undergo adiabatic expansion to be changed into a low-temperature

and low-pressure refrigerant, which is evaporated in an evaporator, and then returned to the compressor. The evaporator to which the low-temperature refrigerant is supplied exchanges heat with air in the compartment, thereby performing cooling.

As the expansion valve, a thermostatic expansion valve is known which senses the pressure and temperature of refrigerant at an outlet of an evaporator, and controls the flow rate of refrigerant supplied to the evaporator such that the refrigerant is in a predetermined degree of superheat (see e.g. Japanese Unexamined Patent Publication No. 2002-310539 (Paragraph Nos. [0034] to [0041], FIG. 6)).

FIG. 7 is a longitudinal cross-sectional view showing an example of the construction of a conventional expansion valve.

The expansion valve 101 includes a body block 102 having side portions formed with a refrigerant conduit connection hole 103 for introducing refrigerant, a refrigerant conduit connection hole 104 for delivering refrigerant, and refrigerant conduit connection holes 105, 106 for being intervened in piping leading from an evaporator to a compressor.

In a fluid passage between the refrigerant conduit connection hole 103 and the refrigerant conduit connection hole 104, a valve seat 107 is integrally formed with the body block 102, and a ball-shaped valve element 108 is disposed in a manner opposed to the valve seat 107 from

the upstream side, and refrigerant undergoes adiabatic expansion when it flows through a gap between the valve seat 107 and the valve element 108. Further, the valve element 108 is urged by a helical compression spring 110 via a valve element receiver 109 for receiving the valve element 108 in a direction of being seated on the valve seat 107. The helical compression spring 110 is received by a spring receiver 111 and an adjustment screw 112.

A power element 113 is provided at an upper end of the body block 102. The power element 113 comprises an upper housing 114, a lower housing 115, a diaphragm 116, and a center disk 117. A temperature-sensing chamber surrounded with the upper housing 114 and the diaphragm 116 is filled with refrigerant, and sealed by a metal ball 118.

The upper end of a shaft 119 is in abutment with the center disk 117. The shaft 119 is inserted through a through hole 120 formed in the body block 102, and has a lower end thereof in abutment with the valve element 108.

The through hole 120 has an upper part thereof expanded, and an O ring 121 is disposed at a stepped portion thereof, for sealing a gap between the shaft 119 and the through hole 120.

Further, the upper end of the shaft 119 is held by a holder 122 which has a hollow cylindrical portion extending downward across a fluid passage communicating between the refrigerant conduit connection holes 105, 106.

The lower end of the holder 122 is fitted in the expanded portion of the through hole 120 and retains the O ring 121.

A coil spring 123 is disposed at the upper end of the holder 122, for suppressing axial vibrations of the shaft 119. The top surface of the holder 122 functions as a stopper defining the maximum valve lift of the expansion valve 101.

In the expansion valve 101 constructed as described above, before the air conditioner is started, the center disk 117 is in abutment with the top surface of the holder 122, and the expansion valve 101 is fully open. Therefore, when the air conditioner is started, the expansion valve 101 starts its operation from the fully open state thereof.

By the way, the automotive air-conditioning system has a different required refrigerating capacity depending on the vehicle to which it is applied, and the capacity demanded of the expansion valve is also different. The capacity of the expansion valve is expressed in tonnage. From the tonnage set depending on the vehicle, the flow rate of refrigerant flowing through the expansion valve is determined, and therefore, the expansion valve is designed such that at least the flow rate corresponding to the set tonnage is guaranteed. In this case, the maximum valve lift is unconditionally set to a sufficiently larger value than that corresponding to the set tonnage, for whatever tonnage the expansion valve may be designed.

However, the conventional expansion valve is fully

open when the air conditioner is started, and the valve lift at that time is larger than that providing a required flow rate, causing a large amount of refrigerant to flow. This increases flow noise generated when the refrigerant passes through the valve, and what is worse, since the expansion valve is excessively opened, this causes the refrigerant to flow at a superfluous flow rate, resulting in an increase in the driving force.

SUMMARY OF THE INVENTION

The present invention has been made in view of these problems, and an object of the invention is to provide an expansion valve that suppresses noise generated during start-up and achieves saving of driving force.

To solve the above problem, the present invention provides an expansion valve including a power element that senses pressure and temperature of refrigerant at an outlet of an evaporator and controls a valve lift of a valve portion, to thereby control a flow rate of refrigerant supplied to the evaporator, characterized in that a maximum value of the valve lift is set such that the flow rate is equal to 1.0 to 1.4 times a flow rate of set tonnage.

The above and other objects, features and advantages of the present invention will become apparent from the following description when taken in conjunction with the accompanying drawings which illustrate preferred

embodiments of the present invention by way of example.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross-sectional view showing an example of the construction of an expansion valve according to the invention.

FIG. 2 is a diagram showing the relationship between the valve stroke and refrigeration ton.

FIG. 3 is a diagram showing the relationship between a factor by which the refrigerating capacity is increased and noise generated during start-up.

FIG. 4 is a diagram showing changes in noise immediately after the expansion valve is started.

FIGS. 5(A) and 5(B) are diagrams for explaining tolerance dispersion, in which FIG. 5(A) illustrates a case of the conventional expansion valve, and FIG. 5(B) illustrates a case of the expansion valve according to the present invention.

FIG. 6 is a longitudinal cross-sectional view showing another example of the construction of the expansion valve according to the invention.

FIG. 7 is a longitudinal cross-sectional view showing an example of the construction of a conventional expansion valve.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereinafter, embodiments of the present invention

will now be described in detail with reference to the drawings.

FIG. 1 is a longitudinal cross-sectional view showing an example of the construction of an expansion valve according to the present invention.

The expansion valve 1 according to the present invention has a body block 2 having side portions formed with a refrigerant conduit connection hole 3 to which is connected a high-pressure refrigerant piping for receiving a high-temperature and high-pressure refrigerant from a receiver/dryer through the piping, a refrigerant conduit connection hole 4 to which is connected a low-pressure refrigerant piping for supplying a low-temperature and low-pressure refrigerant expanded and reduced in pressure by the expansion valve 1 to the evaporator, a refrigerant conduit connection hole 5 connected to refrigerant piping from an evaporator outlet, and a refrigerant conduit connection hole 6 connected to a refrigerant piping leading to the compressor.

Further, in a fluid passage communicating between the refrigerant conduit connection hole 3 and the refrigerant conduit connection hole 4, a valve seat 7 is integrally formed with the body block 2, and a ball-shaped valve element 8 is disposed in a manner opposed to the valve seat 7 from the upstream side. Due to this configuration, a gap between the valve seat 7 and the valve element 8 forms a variable orifice for reducing the

flow of the high-pressure refrigerant, and the high-pressure liquid refrigerant is adiabatically expanded when it flows through the variable orifice. The valve seat 7 is tapered such that the amount of tapering is equal to or more than the amount of axial motion (stroke) of the valve element 8. More specifically, a portion of the valve hole opposed to the valve element 8 has its edge cut to form a tapered hole, and the axial length (height) of the tapered hole is equal to or larger than the length of the stroke of the valve element 8. Here, a smallest-diameter portion of the tapered hole sets the seating position of the valve element 8, and even if the valve element 8 is moved to a farthest position therefrom to make the valve fully open, part of the valve element 8 is positioned within the tapered hole, which prevents the valve element 8 from moving out of the tapered hole when the valve is fully opened.

Further, in the fluid passage on the side of the refrigerant conduit connection hole 3, there are disposed a valve element receiver 9 for receiving the valve element 8, and a helical compression spring 10 for urging the valve element 8 via the valve element receiver 9 in the direction of seating the valve element 8 on the valve seat 7. The helical compression spring 10 is received by a spring receiver 11 and an adjustment screw 12 screwed into the body block for adjustment of load of the helical compression spring 10.

At an upper end of the body block 2, there is provided a power element 13 which comprises an upper housing 14 and a lower housing 15, made of a thick metal, a diaphragm 16 made of a thin metal plate having flexibility and disposed in a manner dividing the space surrounded with the housings, and a center disk 17 disposed below the diaphragm 16. The space surrounded with the upper housing 14 and the diaphragm 16 forms a temperature-sensing chamber which is filled with two or more kinds of refrigerant gas and inert gas, and is sealed with a metal ball 18 by resistance-welding. The center disk 17 has a lower part formed with an increased diameter such that the part radially protrudes outward, and the underside thereof is formed to have a flat surface. The inner wall surface of the lower housing 15 opposed to the underside of a protruding portion of the center disk 17 is also formed to have a flat surface. The flat portion of the inner wall surface functions as a stopper limiting the downward motion of the center disk 17, thereby defining the maximum valve lift of the expansion valve 1.

Below the center disk 17, a shaft 19 is disposed for transmitting displacement of the diaphragm 16 to the valve element 8. The shaft 19 is inserted through a through hole 20 formed in the body block 2.

The through hole 20 has an upper part thereof expanded, and an O ring 21 is disposed at a stepped portion thereof. The O ring 21 seals a gap between the

shaft 19 and the through hole 20, thereby preventing refrigerant from leaking into the fluid passage between the refrigerant conduit connection holes 5 and 6.

Further, the upper end of the shaft 19 is held by a holder 22 which has a hollow cylindrical portion extending downward across the fluid passage communicating between the refrigerant conduit connection holes 5, 6. The lower end of the holder 22 is fitted in the expanded portion of the through hole 20 and the lower end surface restricts the motion of the O ring 21 toward the upper open end of the through hole 20.

A coil spring 23 is disposed at the upper end of the holder 22, for urging the shaft 19 from a radial direction. This configuration of applying lateral load to the shaft 19 with the coil spring 23 prevents the axial motion of the shaft 19 from sensitively reacting to changes in pressure of the high-pressure refrigerant in the refrigerant conduit connection hole 3. That is, the coil spring 23 forms a vibration suppressing mechanism for suppressing generation of untoward vibration noise caused by vibrations of the shaft 19 in the axial direction.

Further, the top of the holder 22 has a passage formed therethrough for communicating the fluid passage communicating between the refrigerant conduit connection holes 5, 6 and the space below the diaphragm 16, and at the same time, the underside of the center disk 17 is formed with a plurality of ventilation grooves in a

radially extending manner, except a central portion with which the shaft 19 is in abutment, thereby allowing the refrigerant returned from the evaporator to enter the chamber below the diaphragm 16.

In the expansion valve 1 constructed as described above, before the air conditioner is started, the power element 13 detects a sufficiently higher temperature than that during operation of the air conditioner, so that the pressure in the temperature-sensing chamber of the power element 13 is made higher, which causes the diaphragm 16 to be displaced downward as shown in FIG. 1, whereby the center disk 17 abuts against the stopper of the lower housing 15. This displacement of the diaphragm 16 is transmitted to the valve element 8 via the shaft 19, thereby making the expansion valve 1 fully open. Therefore, when the air conditioner is started, the expansion valve 1 starts its operation from the fully open state, and therefore, the expansion valve 1 supplies refrigerant to the evaporator at the maximum flow rate.

As the temperature of the refrigerant returned from the evaporator is lowered, the temperature in the temperature-sensing chamber of the power element 13 is lowered, whereby the refrigerant gas in the temperature-sensing chamber is condensed on the inner surface of the diaphragm 16. This causes pressure in the temperature-sensing chamber to be reduced to displace the diaphragm 16 upward, so that the shaft 19 is pushed by the helical

compression spring 10, to move upward. As a result, the valve element 8 is moved toward the valve seat 7, whereby the passage area of the high-pressure liquid refrigerant is reduced to decrease the flow rate of refrigerant sent into the evaporator. Thus, the valve lift of the expansion valve is set to a value dependent on the cooling load.

FIG. 2 is a diagram showing the relationship between the stroke of a valve and refrigeration ton.

The expansion valve 1 has its capacity determined according to the refrigerating capacity demanded by the system, and in general, there are a 1.0-ton type, a 1.5-ton type, and a 2.0-ton type of expansion valves. In all of these types, the valve element 8 has its valve lift controlled within a range of stroke corresponding to the associated refrigeration ton. For conventional expansion valves, the maximum valve lift during start-up is set, irrespective of the type, by a certain sufficiently large value of stroke A, e.g. 0.8 mm. However, in the expansion valve 1 according to the present invention, the maximum valve lift is set by such a stroke as will cause the refrigerant to flow at 1.0 to 1.4 times the flow rate corresponding to the designated tonnage. For example, in the case of the 1.0-ton type expansion valve, the maximum stroke is set to a value within a range between a stroke position B allowing the refrigerant to flow at a flow rate satisfying the capacity of 1 ton and a position B′ of the maximum valve lift allowing the refrigerant to flow at 1.4

times the above flow rate.

FIG. 3 is a diagram showing the relationship between the scale factor of the refrigerating capacity and noise generated upon start-up, and FIG. 4 is a diagram showing changes in noise immediately after the operation of the expansion valve is started.

FIG. 3 shows how the noise generated during start-up of the expansion valve 1 varies with a change in the scale factor of the refrigerating capacity. According to FIG. 3, noise is steeply increased when the refrigerating capacity is in the vicinity of 1.4 times or exceeds the same. The expansion valve is configured such that the refrigerating capacity can only be increased by a factor of 1.4 at the maximum, in the fully-open state of the expansion valve, which makes it possible to suppress generation of noise during start-up.

Further, due to the refrigerating capacity limited up to a scale factor of 1.4, the noise immediately after start-up is made much smaller than the prior art, as shown in FIG. 4. With the lapse of time, the refrigeration cycle becomes stable, causing the expansion valve 1 to enter the control region, so that the noise becomes equal in magnitude to the prior art.

FIGS. 5(A) and 5(B) are diagrams for explaining tolerance dispersion, and FIG. 5(A) shows a case of a conventional expansion valve, while FIG. 5(B) shows a case of the expansion valve according to the present invention.

The expansion valve according to the present invention is required to make the maximum stroke of the shaft smaller than that of the conventional expansion valve. For example, in the case of 1.0-ton type expansion valve, the maximum stroke of the shaft is reduced from the conventional value of 0.8 mm to a value of 0.3 mm. Therefore, tolerance dispersion in the sizes of members determining the stroke has a large influence on the valve, and therefore the dispersion is required to be made small. The expansion valve according to the present invention solves this problem by changing the stopper of the center disk 17 from the holder 22 to the lower housing 15 of the power element 13.

More specifically, in the conventional expansion valve, as shown in FIG. 5(A), the stroke S of the shaft 119 is from a position at which the center disk 117 is in contact with the top surface of the holder 122 to the illustrated position assumed when the valve is fully closed. Further, P designates an amount of protrusion of the shaft 119 from the top surface of the body block 102 when the valve is fully closed. Further, A designates a height from the stepped portion of the body block 102, where the holder 122 is received, to the top surface of the body block 102, B a height from the bottom surface of the holder 122 lying on the stepped portion to a surface of the holder 122 with which the center disk 117 is brought into contact when the valve is fully opened, and C

a height from the surface of the center disk 117 with which the shaft 119 is in abutment to a surface of the center disk 117 which is brought into contact with the holder 122 when the valve is fully opened.

By using the stepped portion receiving the holder 122 of the body block 102 as the reference, an expression of $(A + P) + C = B + S$ holds, and from this, the stroke S can be expressed as $S = A + P + C - B$. More specifically, the number of parameters determining the stroke S is four.

On the other hand, in the expansion valve 1 according to the present invention, as shown in FIG. 5(B), the stroke S of the shaft 19 is from a position in which the center disk 17 is in abutment with the inner surface of the lower housing 15 to the illustrated position assumed when the valve is fully closed. Now, if the top surface of the body block 2 on which the power element 13 is mounted is used as the reference surface, and the thickness of the lower housing 15 is represented by t, the amount P of protrusion of the shaft 19 from the reference surface is expressed as $P = t + S$, so that the stroke S can be expressed by $S = P - t$. Therefore, the number of parameters determining the stroke S becomes two, which means the number of dispersion-causing factors is reduced to half. This makes it possible to make the tolerance dispersion smaller than the conventional expansion valve.

Particularly, when the holders 22, 122 are made of resin, since the resin is thermally expanded, the

conventional expansion valve suffers from dispersion of the parameter B caused by the refrigerant temperature, which makes the value of the stroke S a function of temperature. In contrast, in the present expansion valve, the parameters determining the stroke S do not contain the parameter B, which makes it possible to further decrease the tolerance dispersion.

FIG. 6 is a longitudinal cross-sectional view showing another example of the construction of the expansion valve. It should be noted that in FIG. 6, the component elements identical to those shown in FIG. 1 are designated by the same reference numerals, and a detailed description thereof is omitted.

An expansion valve 1a according to this embodiment is different from the expansion valve 1 shown in FIG. 1 in which the center disk 17 is guided by the inner wall surface of a vertical portion of the lower housing 15, in that the same is guided by the holder 22 of the shaft 19.

More specifically, the center disk 17 has a lower central portion protruding downward, and this protruding portion is inserted in a hole formed in the top of a holder 22, whereby it is guided by the holder 22 in a manner movable forward and backward along the axis of the shaft 19. This causes the center disk 17 to be positioned by the holder 22 on the same axis as that of the shaft 19, which enables the center disk 17 to move smoothly without being caught by the lower housing 15 when the center disk

17 is moved forward and backward by displacement of the diaphragm 16, providing stable flow rate characteristics.

The center disk 17 is configured such that a surface of the protruding portion for abutment with the shaft 19 and a surface thereof for abutment with the stopper of the lower housing 15 are both formed to be flat, and a plurality of ventilation grooves is formed on the surface for abutment with the stopper in a radially extending manner, thereby allowing the refrigerant returned from the evaporator to enter the chamber below the diaphragm 16 via the ventilation grooves even when the valve is in the state of the maximum valve lift in which the center disk 17 is brought into contact with the lower housing 15.

Further, the expansion valve 1a according to this embodiment is configured such that an adjustment screw 12a also plays the role of the spring receiver to thereby reduce the number of component parts.

As described heretofore, the expansion valve according to the invention is configured such that the maximum valve lift provides a flow rate of refrigerant which is 1.0 to 1.4 times the flow rate corresponding to the set tonnage. This limits the flow rate of refrigerant when the valve is fully open during start-up, thereby enabling reduction of noise generated when the refrigerant passes through the valve, and prevents an unnecessarily excessive amount of refrigerant from flowing, thereby enabling prevention of wasteful use of driving force.

Further, the stroke of the center disk of the power element toward the valve portion is restricted by the inner wall of the valve portion-side housing. This reduces the number of parameters determining the stroke, which makes it possible to make the number of factors causing tolerance dispersion of the valve stroke smaller than the prior art.

Further, the valve seat is tapered, and the length of the tapered portion in an axial direction is made equal to or more than the length of stroke of the valve element. This makes it possible to prevent the valve element from moving out of the tapered hole even when the helical compression spring urging the valve element is in an inclined state.

The foregoing is considered as illustrative only of the principles of the present invention. Further, since numerous modifications and changes will readily occur to those skilled in the art, it is not desired to limit the invention to the exact construction and applications shown and described, and accordingly, all suitable modifications and equivalents may be regarded as falling within the scope of the invention in the appended claims and their equivalents.